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AN INNOVATIVE ELECTRIC SHIP STEERING SYSTEMS: ANALYSIS AND IMPLEMENTATION

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ABSTRACT

Ship's steering systems have changed over the ages, driven by the need for more powerful and reliable systems. Nowadays the normal practice for ship's steering systems lies into hydraulic solutions because of their excellent characteristics in terms of performances and reliability. However, the modern advancements in the electric drives field, give insights to compare the two systems, with attention to the lower masses and costs enabled by electric drives adoption. This paper describes this comparison, downstream to the conceptual design purpose of an electric drive system suited for ships of about 500 tons of load displacement and 50 m of overall length. The results confirm the hypothesis, laying the foundations for the development of ship's steering systems inspired by this studied solution.

KEYWORDS: Ship Steering System, Steering Gear, Brushless Motor & Differential Gear Drive

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Original Article

INTRODUCTION

The rudders drive system issues have a very long history that basically starts since the first ships development. At the very beginning these systems were manual, then electric motors and hydraulic systems have been introduced; however, in the last decades the great majority of ship's designers have shifted to the sole hydraulic solution, because of their better reliability and performances with approximately the same dimensions and masses. Despite this trend, it is worth making some considerations on the modern opportunities offered by the new electric motors and electronic controllers, primarily in terms of mass and cost reductions [1]. To develop these considerations into details, a case study has been considered and will be described in the next sections.

The approach followed is part of a more general trend that, in recent years, has seen a growing interest in replacing traditional solutions with more innovative ones that arise from the integration of special 3D printed components integrated into mechatronic systems [2], [3].The implementation of the electric drive system for the rudders must ensure the performances and the reliability according to the regulations of the American Bureau of Shipping (ABS), which will be considered as the reference also for the load calculations [4], [5].

The comparison starts from the case study description, then calculations for the load determination have been reported, together with some hypothesis on the rudders motion law. Then a complete solution for the electric drive is purposed, including the choices of the mechanical transmission, the gear box, the electric motor and an original differential gear drive.

Finally, a comparison based on costs, masses and occupied volumes, is reported. The results highlight the convenience, in that terms, of the electric drive solution compared with the hydraulic one.

Application Description

The characteristics of the naval units under consideration within this study are synthesized in table 1.

Table 1: Characteristics of the Naval Units under Consideration

Length overall	~ 55 m
Length between perpendiculars	~ 50 m
Breadth (moulded)	~ 8.5 m
Depth	~ 5.5 m
Full load displacement	500 t
Corresponding moulded draught	2.4 m
Maximum continuous speed	21 knots
Rudder profile type	Gottingen
Rudder section (trapezoidal)	2 m^2
Mass (suspended part)	885 kg
Moment of inertia	62.9 kg · m^2

The main requirements, according to [4], can be summarized as follows:

- "The main steering gear is to be capable of putting the rudder from 35° on side to 35° on the other side with the vessel running ahead at maximum continuous shaft rpm and at the summer load waterline; and, under the same condition, the travel time from 35° on either side to 30° on the other side is not to be more than 28 seconds".
- "An auxiliary steering gear is not required under the following conditions. When the main steering gear comprises two or more power units, and is so arranged that after a single failure in its piping system or in one of the power units the defect can be isolated so that the steering capability can be maintained or regained; and provided that (a) for passenger ship, the main steering gear is capable of operating the rudder while any one of the power units is out of operation; and (b) for cargo ship, the main steering gear is to be capable of operating the rudder while all the power units are in operation".

According to the aforementioned requirements, it is allowed to have one main steering system fed by two different power systems, both able to guarantee the required performances independently. Considering the similar costs, and in order to extend the maintenance intervals, it is convenient to have two motors of the same size.

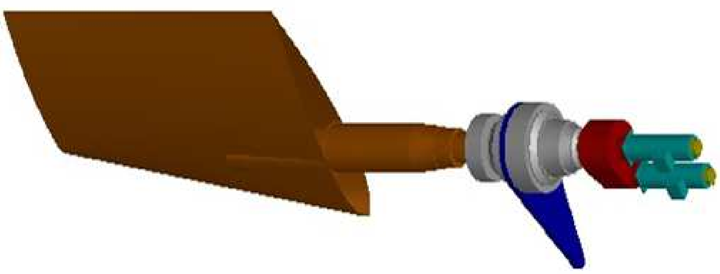

Figure 1: Trapezoidal Rudder with Gottingen Profile

Load Determination

The force acting on the rudder is estimated, according to [6], with the following formula, calculated in the most onerous conditions (rudder angle of 35°):

$$T_R = n \cdot K_T \cdot A \cdot V^2 \tag{1}$$

Where T_R indicates the rudder torque, $A = 2\ m^2$ is the total projected rudder area, V = 21 kts is the maximum design ahead speed, $K_T = 1.463$ is the ahead motion coefficient and n = 0.132 is the profile coefficient.

Therefore, the maximum torque on the rudder's rod is calculated as:

$$T_{r\,max} = T_R \cdot r = n \cdot K_T \cdot A \cdot V^2 = 19.6\ kN \cdot m \tag{2}$$

Where $r = 0.1 \cdot c = 0.115\ m$ indicates the distance of the point of application of the pressure resultant force on the rudder blade from the rudder's rotational axis.

The engine shaft's average angular speed, to fulfill the requirements of [5], is calculated as following:

$$\omega_r = n_r \cdot \frac{2\pi}{60} = 0.0406\ rad/s \tag{3}$$

Where the rudder rotational speed is:

$$n_r = \left(\frac{\varphi}{360}\right) \cdot \left(\frac{60}{t}\right) = 0.387\ rpm \tag{4}$$

The load characteristic in the $T_r - \omega_r$ diagram is reported in Figure 2.

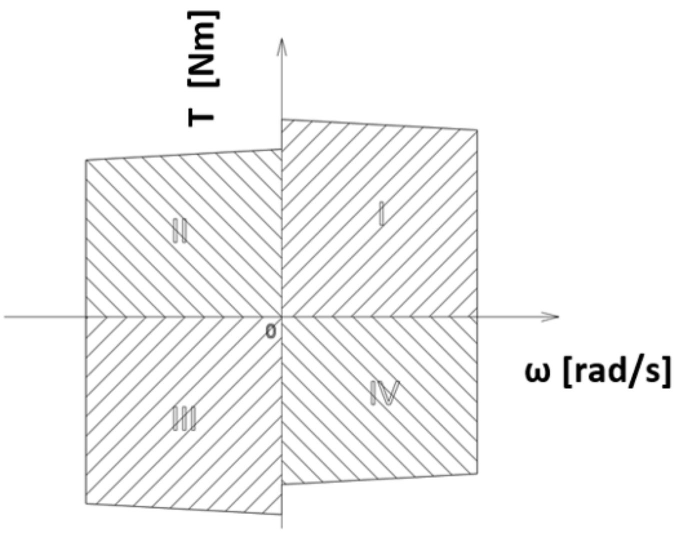

Figure 2: Load Characteristic

The first and the third quadrant represent a passive load, therefore the rudder is increasing its incidence angle; on the contrary the second and the fourth quadrant represent an active load, therefore the rudder is decreasing its incidence angle. The step torque at the load inversion ($\omega_r = 0$) is due to friction and is quantifiable in approx.1500 $N \cdot m$. The maximum torque is reached at maximum rudder's incidence angle (35°) at $\omega_r = 0$.

The power that the engine needs is:

$$P_r = T_{r,max} \cdot \omega_r = 800W \quad (5)$$

Motion Law

In order to define a motion law, the minimum kinematic parameters have been defined; in particular these parameters are summarized in Figure 3.

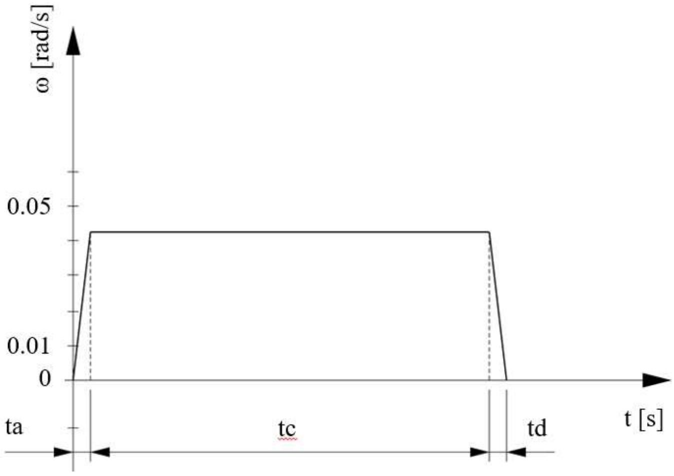

Figure 3: The Law of Motion

The acceleration time (ta in figure) has been set to 1s; the remaining time is passed at constant speed for 27 s until the overall length of the rudder's motion is reached (from 35° to -30°).

The minimum load's angular speed required to fulfil the performance requirements is:

$$\omega_{r,min} = \frac{\Delta\varphi}{\left(\frac{ta}{2}+tc\right)} = 0.0413\, rad/s \quad (6)$$

As a consequence, the angular acceleration results $\dot{\omega} = 0.0413\, rad/s^2$, being $\Delta\varphi = 1.135\, rad$

Once the minimum kinematic parameters have been determined, some standardized tests [7] and [8] can be applied to evaluating rudder performances:

- The ship, while it is going at its maximum speed, is asked for a sharp turn of the maximum rudder angle (35°) maintained for a complete 720° turnaround. During this test some values are registered and should be compliant with the standards.
- The ship is asked for a zig-zag trajectory at its maximum speed, with rudders movements of 40°, from 20° on one side, to -20° on the other one; this repeated for at least five cycles.

These tests are important rules to be taken into account during the steering system design; in particular an hypothetic motion law, containing the requirements expressed by the tests, has been issued and reported in Figure 4, where the dashed line considers friction, and it is approximated with the continuum one, which, neglects the phenomena. The motion law represents a hybrid of the capabilities that are required by [7] and it is basically a cycle of 0°→35°→pause→-35°→0° and so on.

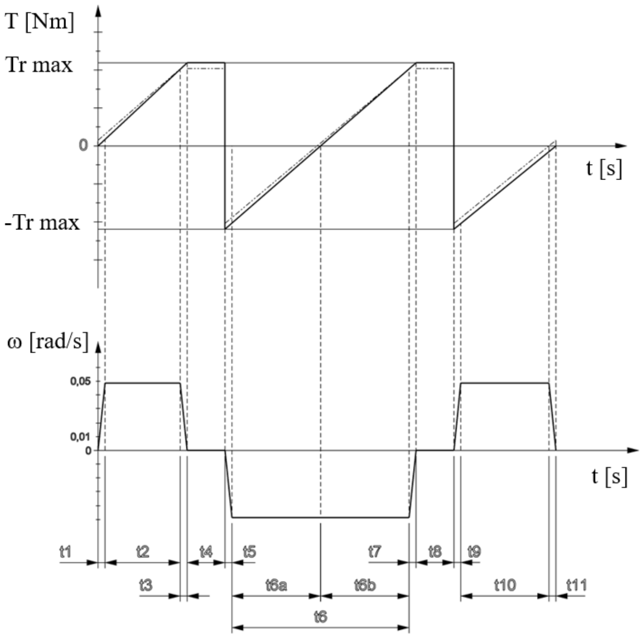

Figure 4: Duty Cycle

Choice of Construction Details

The overall block schema for the steering system purpose is reported in the figure below; then each single choice will be discussed in the next sections.

The first components to be analysed are the motors: this application is characterized by relatively low velocities, accelerations and load inertia, therefore expensive motors with high dynamic performances are not required [9]. Thus, the choice is driven by costs reduction, and fell on DC brushed motors.

The gearbox has been chosen in order to guarantee reliability and ease of maintenance, moreover, the application does not require a high transmission ratio, therefore an adequate solution is a planetary shaft mounted gearbox [10].

The differential gear drive is a specialized device seldom adopted and not available on the market in standardized range. This lack leads to the necessity of bespoken design [11] to address the application requirement [12]. The gears of the combiner have been sized according to good mechanics design practice and the professional experience of the authors. The verification has been carried out according to [13] to be able to extend this procedure to all the products subjected to the supervision of the various competent Naval Registers.

Transmission shafts have been sized and verified, bearings have been sized and verified on the basis of the maximum external loads imposed on the inputs and output shafts.

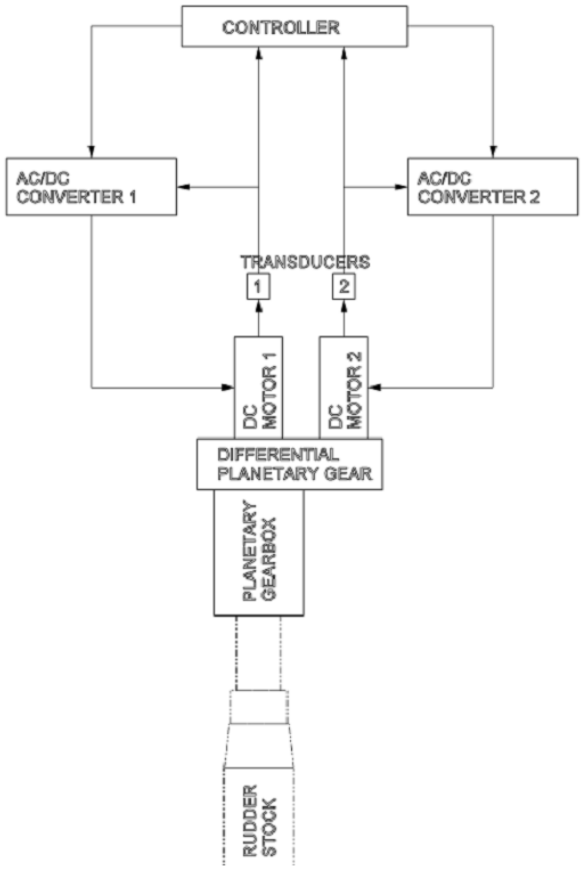

Figure 5: The Block Diagram of the Overall Steering System

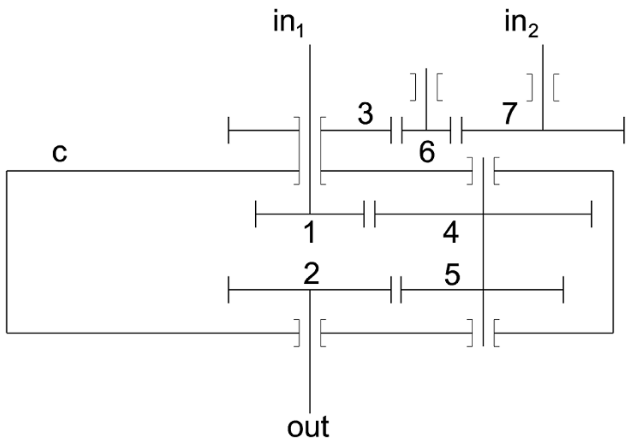

Figure 6: Diagram of the Differential Gear Drive

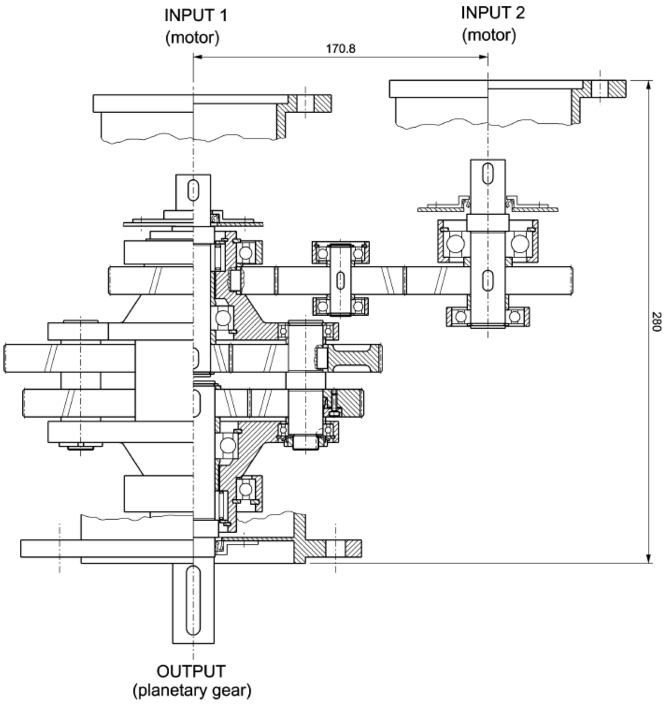

Figure 7: CAD Drawing of the Differential Gear Drive

CONCLUSIONS

The electric servo drive allows easy manoeuvring of the rudders and has been designed in all its parts with satisfactory results. The overall system is very simple, ordinary maintenance is limited to the engines and is not expensive. In the event of an engine failure, the replacement becomes a quick and simple operation, such that it can also be carried out while sailing.

If changing the overall dimensions of the servo drive is needed, it is possible to select a gearbox in an angular version, i.e. the input axis is orthogonal to the output, orienting the differential drive unit parallel to the reaction arm.

The presence of the differential drive is fundamental to guarantee the regulatory requirements. It allows to double the speed performances and it gives possibility to perform the instantaneous change of the motor without carrying out particular manoeuvres but simply power supplying one or the other, providing performances much closer to those of the alternative hydraulic systems.

The electric drive has lower overall masses and volumes than the hydraulic system with linear actuators and similar to the compact, but expensive, semi-rotary actuators with integrated control unit. Moreover, it is always less expensive, and it allows to eliminate the fluidic lines needed by hydraulic systems to connect the control unit and actuators with oil pipes and to supply sea water to the heat exchangers for cooling the hydraulic oil. Furthermore, a drastic reduction in the quantity of used oil make it even more environmentally friendly. The principles of the present study are easily extendable to rudders with resistant torque up to 50 KN but by installing brushless motors rudders can be operated with resistant torque up to 600 kN (limit dictated by the transmission system).

Table 2: Mass Comparison

Subsystem	Hydraulic Mass (kg)	Electric Mass (kg)
Centraline	270	-
Hydraulic cylinders	240	-
Rudder	231	-
Gearboxes	-	400
Differential gear drive	-	60
Motors	-	60
Reaction arms	-	25
Motor panels	128	46
Transformers	-	120
Compensator	28	-
Oil reservoir and pump	60	-
Control system	33	33
Oil	280	16
Cables	120	120
Oil pipes	50	-
Water pipes	70	-
Total	**1510**	**880**

Table 3: Cost Comparison

Subsystem	Hydraulic Cost ($)	Electric Cost ($)
Hydraulic system	164000	-
Gearboxes	-	11000
Differential gear drive	-	13800
Motors	-	6500
Encoders	-	1100
Reaction arms	-	970
Converter and panel	-	5660
Transformers	-	2500
Control system	-	3500
Oil	1000	130
Oil pipes	9000	-
Water pipes	9650	-
Indirect costs - management	-	13500
Total	**183650**	**58660**

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